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A STUDY OF THE EFFECT OF LUBRICATION ON
THE DYNAMICS OF SPINNING SPINDLES

Crawford

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A STUDY OF THE EFFECT OF LUBRICATION ON
THE DYNAMICS OF SPINNING SPINDLES

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A STUDY OF THE EFFECT OF LUBRICATION ON THE DYNAMICS OF SPINNING SPINDLES

INTRODUCTION

Successful and economical spinning operations depend to a large extent on the satisfactory performance of spindles. Despite the introduction of numerous refinements, the principal parts remain analogous to the ones used in the nineteenth century, yet the modern trend toward high spindle speeds and large packages has aggravated the interrelated problems of power consumption, vibration, and lubrication. Investigations on these problems have been made in recent years by several authors or organizations.¹ These studies, of value to mill operations, were conducted chiefly on the existing spindle designs by the suppliers of lubricants, and the information furnished is experimental only. In regard to work on actual design, attention has been concentrated on the dynamics and balancing without due consideration of the effect of the lubricants involved.²

1. "Textile Machinery Lubrication", Lubrication, Vol. 16, pp. 13-24, 1930. "Spindle Design and Function of Spindle Oils", Lubrication, Vol. 23, pp. 25-31, 1937. "Vibrating Spindles: Causes and Cure", Textile World, Vol. 89, pp. 76-7, 1939. F. S. Jones and S. P. Marley, "Textile Spindle Performance Characteristics", presented at annual meeting of A. S. L. E., April, 1948.

2. Y. Sakurai, "The Study of Form to Decrease the Vibration of High Speed Spindle in Cotton Spinning Machine", Soc. of Mech. Engrs., Japan -Transaction, No. 9, pp. 529-31, 1936.

The purpose of this study was to set an example for the combination of a theoretical study of dynamics as involved in fundamental design and the evaluation of experimental results in the interrelated factors such as speeds, vibration, lubrication, and power consumption, the study of oil throwing being secondary. The experiments were made on sleeve bearing spindles which are now extensively used in mills.

EXPERIMENTAL WORK

Experiments were made on a modified long-draft spinning frame of 36 spindles, set for producing 22s warp yarn from 4-hank roving of middling cotton so as to obtain actual operating conditions. A continuous run of two doffs was made with the spindles running at speeds of 4,100, 6,900, 8,100, 9,600, and 10,600 R.P.M. with each of the six grades of oil of different viscosity as listed in Table I. The letter designations from this table are employed in all of the diagrams, for simplicity. In Figs. 1, 2, 3, and 4, the general appearance of the apparatus is shown.

Power was measured with a dynamometer equipped with ball-bearing axles on which the driving motor was mounted, as shown in Fig. 1. Readings of the dynamometer were recorded (summarized in Table II on page 44), on the following operations: (a) Running the drafting system at various speeds, (b) Running the drafting system and spindles at various speeds, and (c) Running the drafting system and spindles spinning at various speeds. Therefore, from these data, the power for driving an individual spindle was calculated, with the secondary purpose of showing the power distribution on the spinning frame (Fig. 5). The power-speed curves in Fig. 6 were obtained from those in Fig. 5, converted to watts in order to get a larger scale for comparison. A constant check and recording of spindle speeds, cylinder speeds, and motor speeds was made by using the strobotac shown in Fig. 4. The

TABLE I
VISCOSITY OF SPINDLE OILS USED IN EXPERIMENT ³

Designation	Saybolt Universal Viscosity			
	70°F	100°F	130°F	210°F
A	103.7	63.5	48.2	35.3
B	155.0	83.6	57.5	37.9
C	240.0	109.9	68.3	40.0
D	300.0	131.2	77.0	41.6
E	329.0	143.5	82.3	42.4
F	452.0	186.5	98.0	45.2

³. Courtesy of the Texas Co.

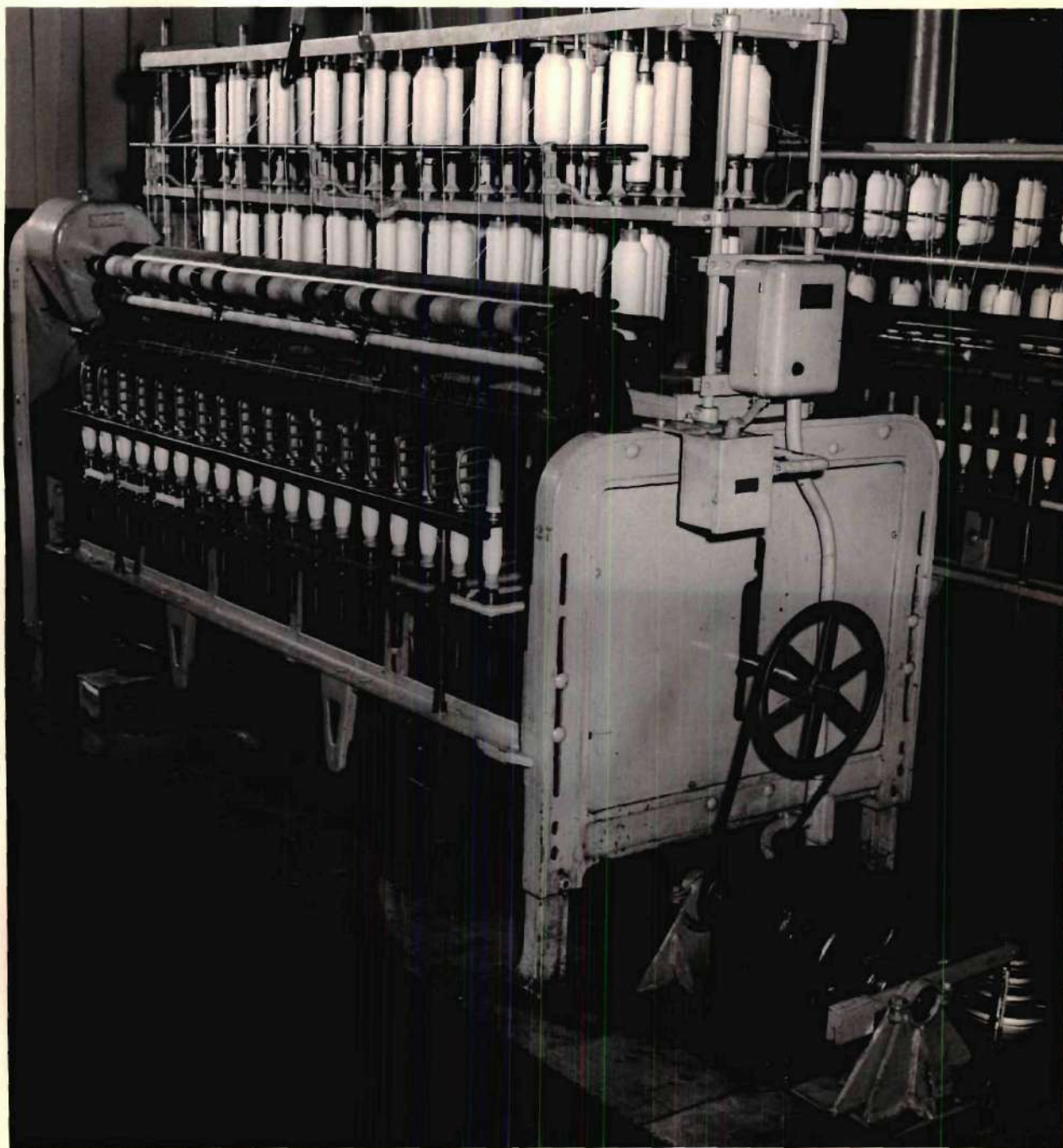


FIG. 1 SPINNING FRAME AND DRIVE, WITH DYNAMOMETER
AND CHANGE PULLEYS

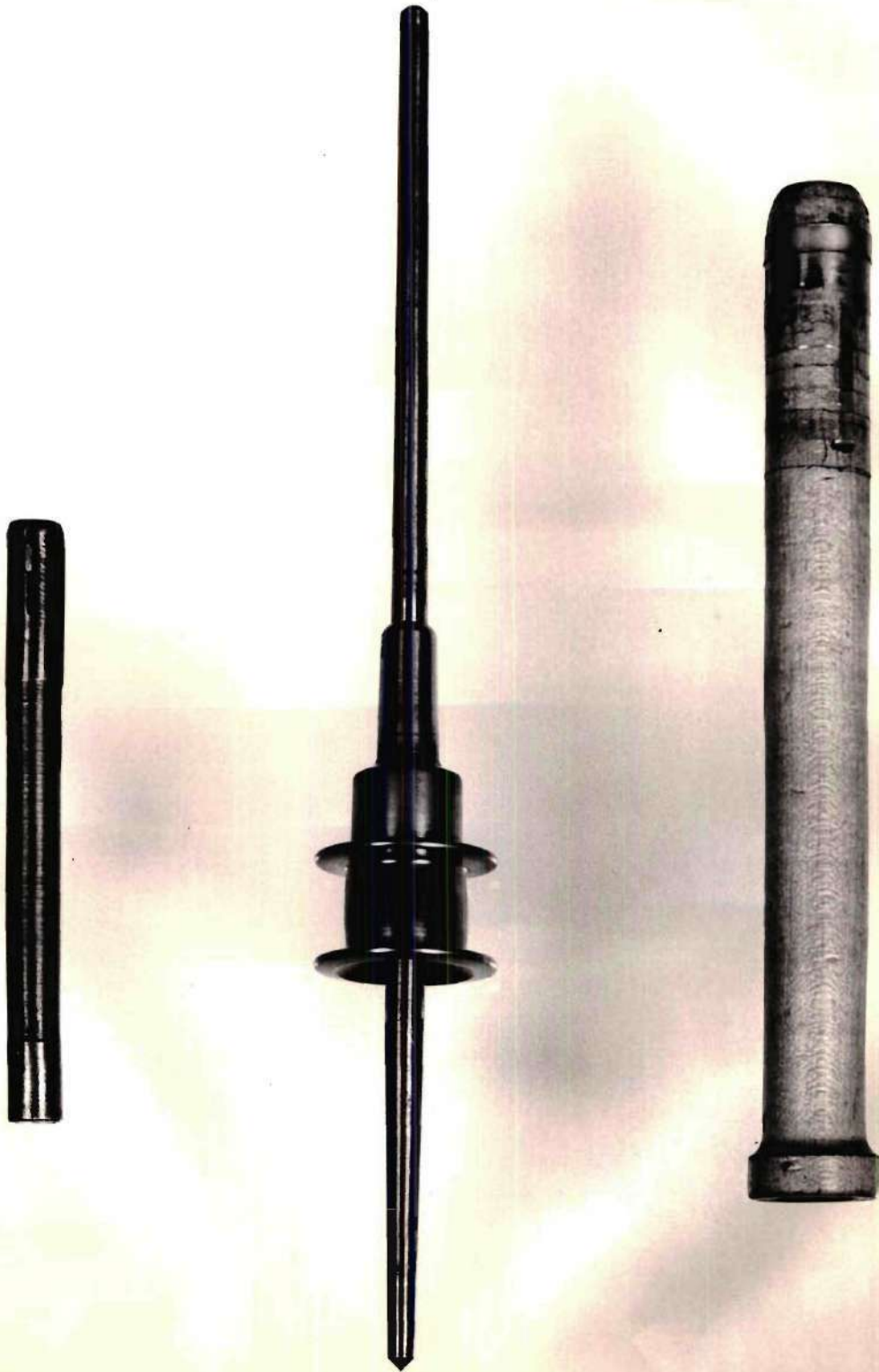


FIG. 2 SPINDLE, BOLSTER, AND MODIFIED BOBBIN
FOR VIBRATION MEASUREMENT

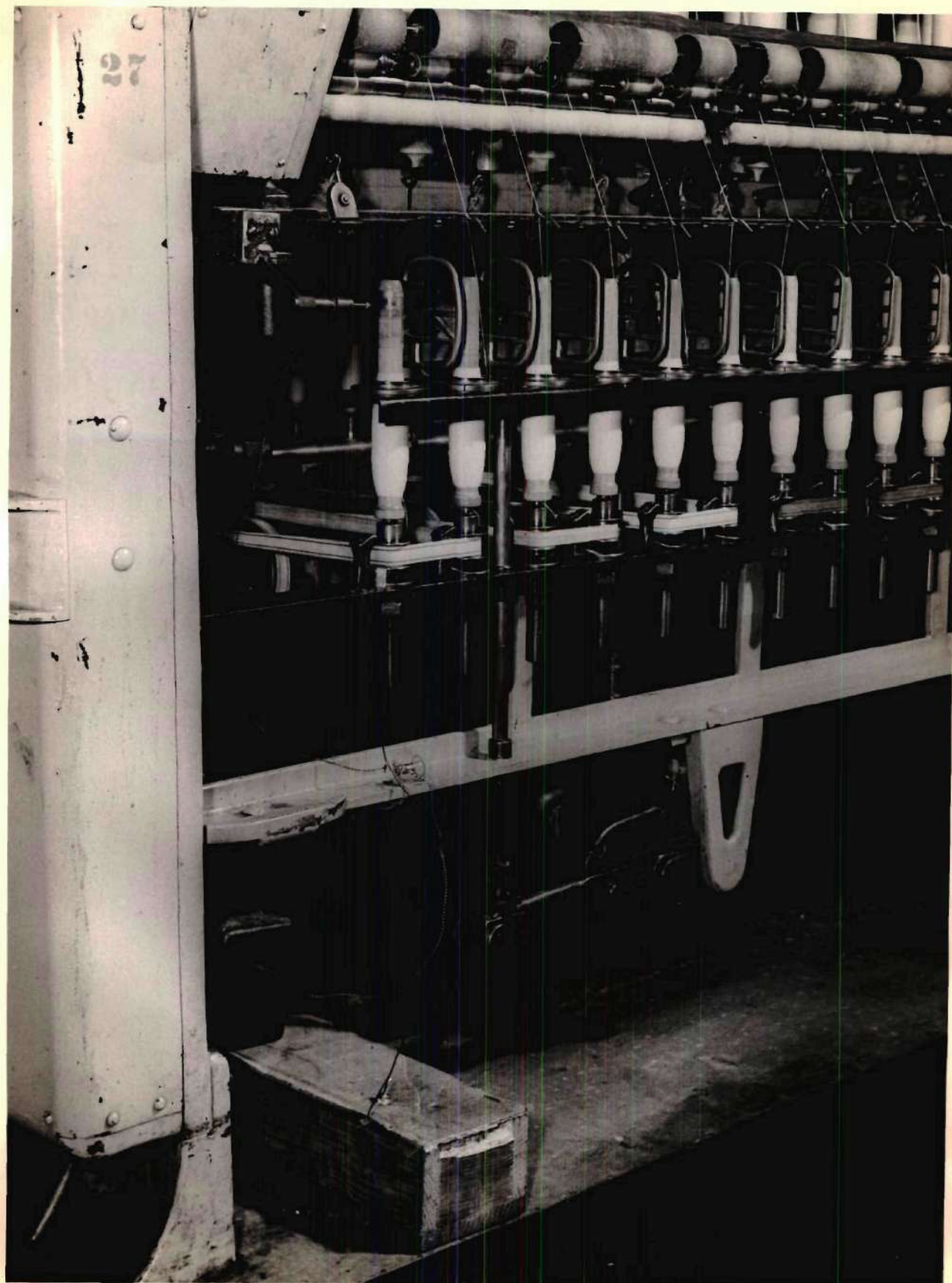


FIG. 3 ELECTRIC CIRCUIT AND MICROMETER USED
FOR MEASURING VIBRATION OF SPINDLE

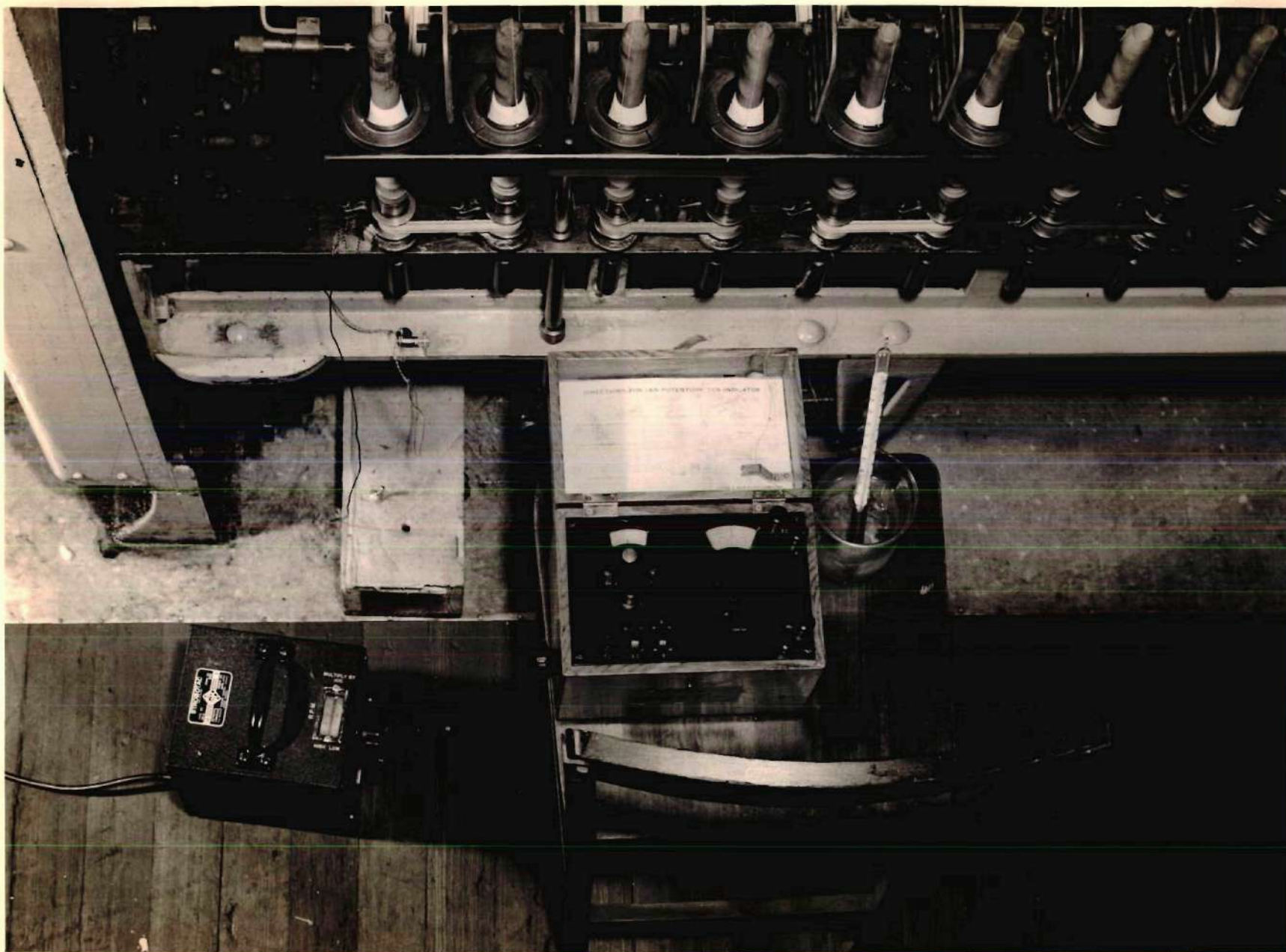


FIG. 4 THERMOCOUPLE CONNECTED WITH POTENTIOMETER FOR MEASURING OIL TEMPERATURE AT SPINDLE BASE. STROBOTAC AT LEFT WAS USED FOR MEASURING SPINDLE SPEED

average percentage slip of the spindle drive was plotted as shown in Fig. 7.

The amplitude of vibration or wobble (see page 29) was measured by an electric circuit and micrometer, as shown in Fig. 3. Measurements were made by electrical contact between the micrometer point and a metal collar mounted on the tip of the bobbin. The metal collar of thin brass sheet was grounded on the spindle. Electric current was transmitted from the spindle to the metal brush located on the bolster rail and, in turn, to the indicating bulb. Readings were taken with the spindle rotating at specified speeds and at rest, and the difference between these two conditions was taken as the amplitude at the tip. Data were taken at empty bobbin, at half full bobbin, and at full bobbin. The spindle located at another position on the bolster rail was transferred to the position shown in Fig. 3 to get average values of amplitude of vibration. The same bobbin was used for all measurements. Diagrams were plotted from the data obtained, in Figs. 9 to 14 (inclusive).

The temperature of the bearing oil was measured near the end of each test run by a copper-constantin thermocouple connected with a potentiometer, as shown in Fig. 4. The average temperature rise from room temperature (80 to 95°F, as recorded) was plotted against spindle speed for different oils, as shown in Fig. 8.

Sample bobbins were taken after each doff and observed under ultraviolet light for oil stains. Oil throwing was quite evident at spindle speeds of 6,900 R.P.M. and up with various oils when the oil well was first filled. The net effect was to make the floor and bolster rail oily. Only negligible oil staining of the yarn was observed with oil lighter than 109.9 S.U.V. at 100°F and at spindle speeds above 8,100 R.P.M. While oil throwing was observed during the first hour after filling the oil well, no throwing was observed after the oil level dropped to a certain height, this height being higher for oils of higher viscosity and lower for oils of lower viscosity.

The eccentricity of the assembly of yarn, bobbin, and spindle was obtained by balancing this assembly along a pair of tempered steel straight edges which was well levelled, using pieces of wax stuck over the outer surface of the bobbin to obtain a complete static balance. This wax was then carefully removed and weighed, and the eccentricity was calculated by use of the following equation:

$$M \cdot e = m \cdot r,$$

where M is the weight of the whole assembly, e is the eccentricity, m is the weight of wax required for balancing, and r is the radius of the bobbin. In this experiment, M was 225.35 grams, r was 11.75 mm., and the average weight of m was found to be 1.014 grams. Therefore, e was equal to 0.053 mm.

POWER-SPEED CURVES OF THE WHOLE FRAME

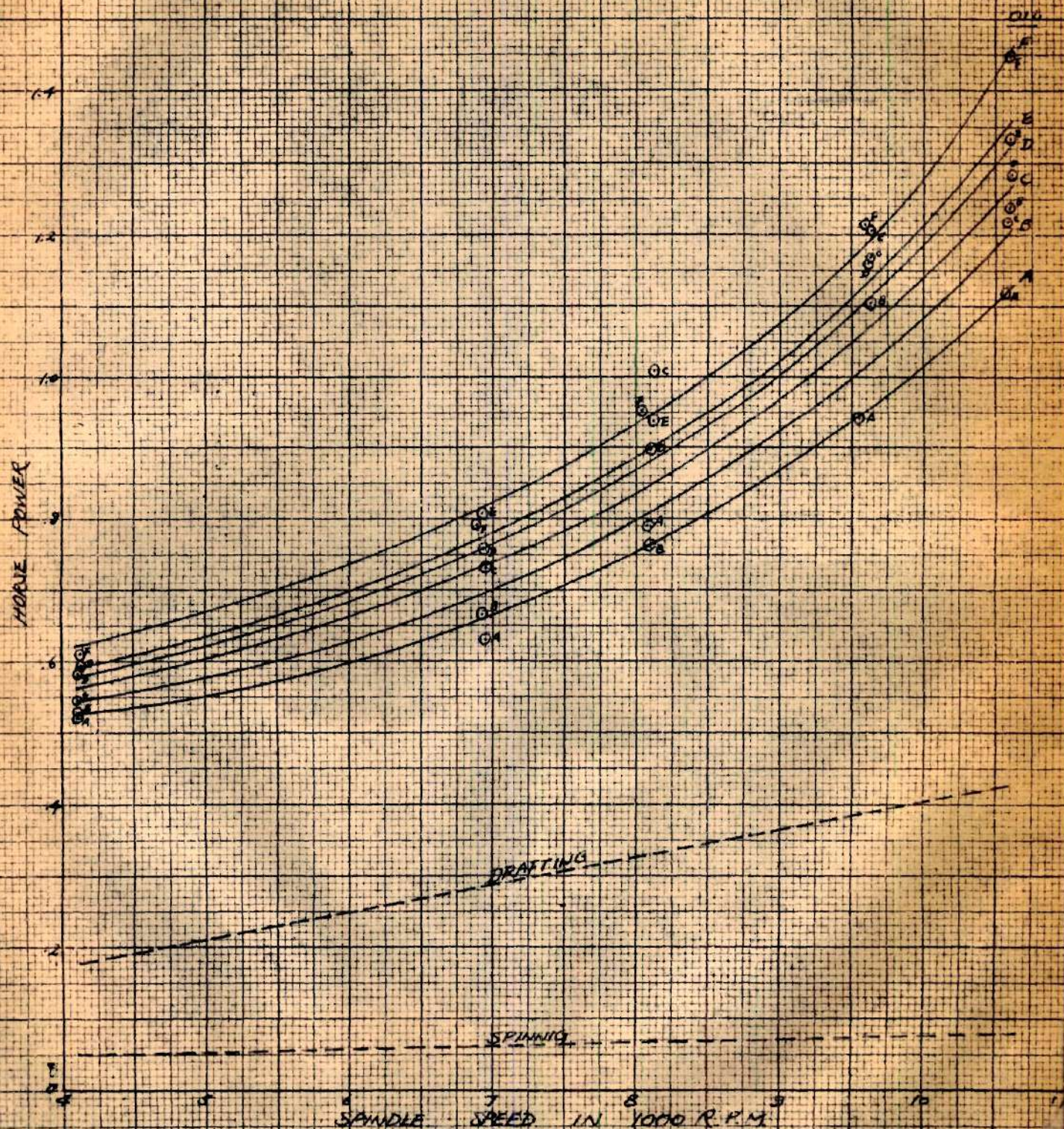


FIG. 5

POWER-SPEED CURVES
SINGLE SPINDLE
(CALCULATED FROM CURVES IN FIG. 5)

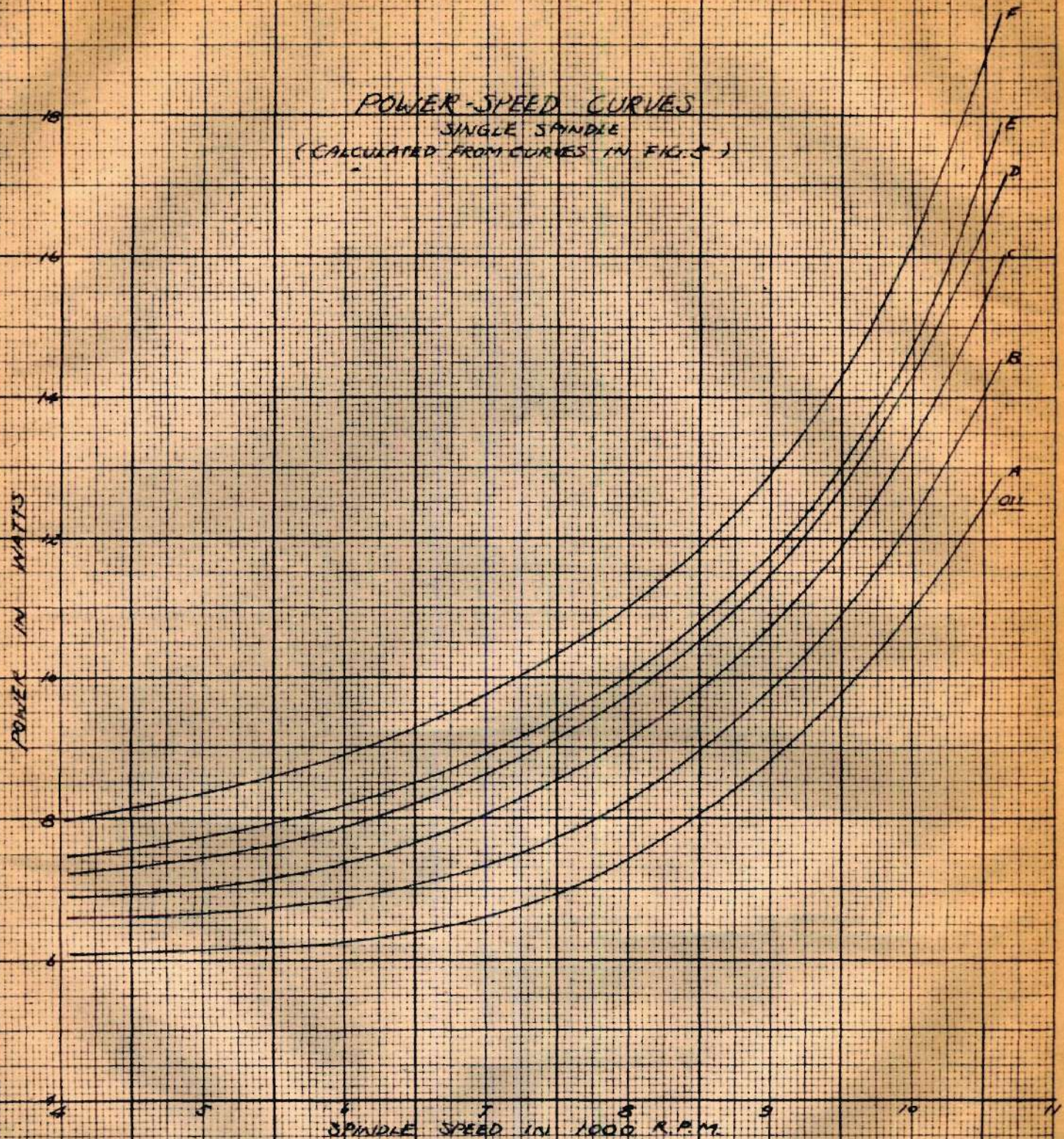


FIG. 6

PERCENT SLIP OF TAPE DRIVE

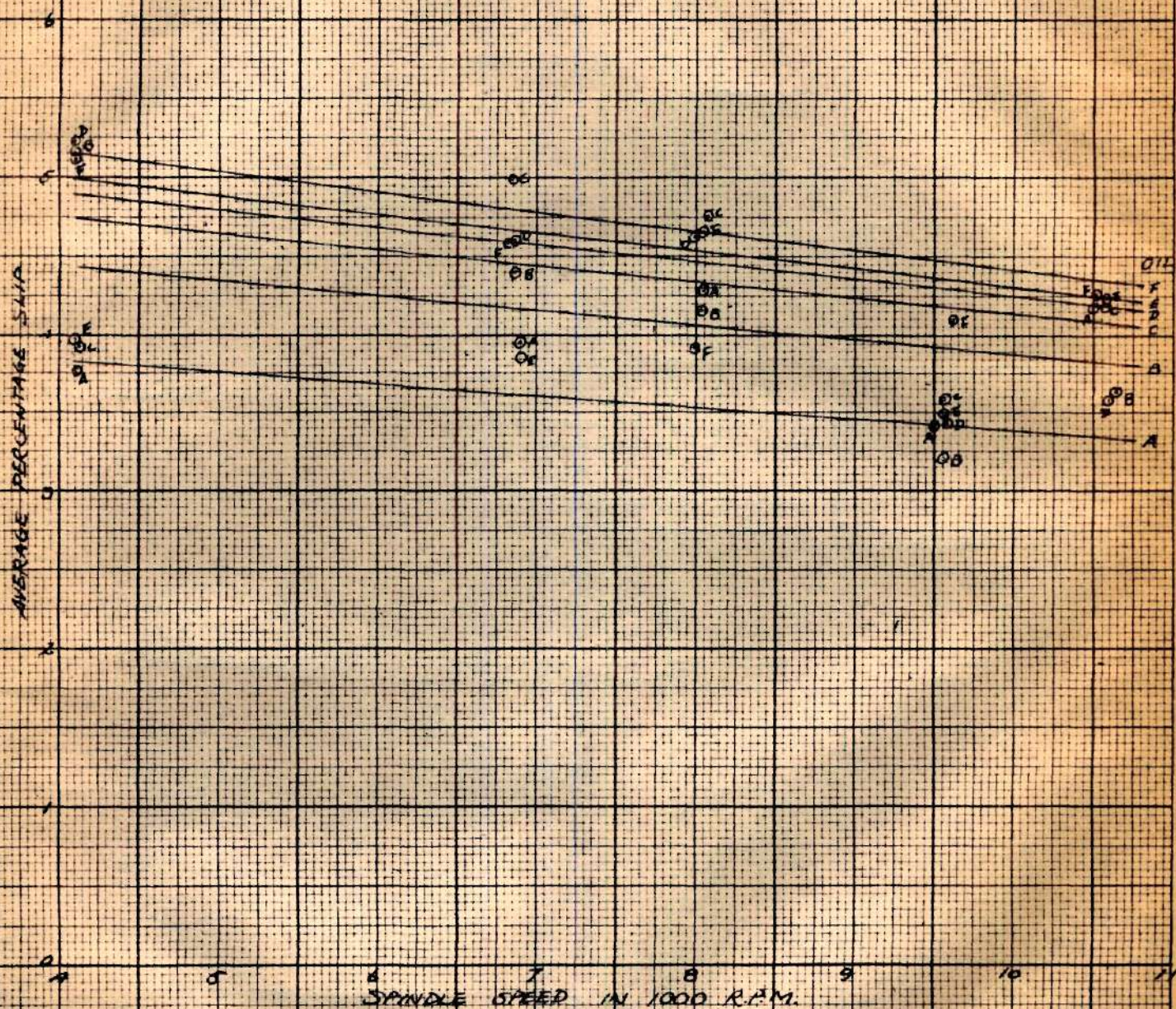


FIG 7

AVERAGE BEARING TEMPERATURE RISE
(ROOM TEMP. 80 TO 90°F)

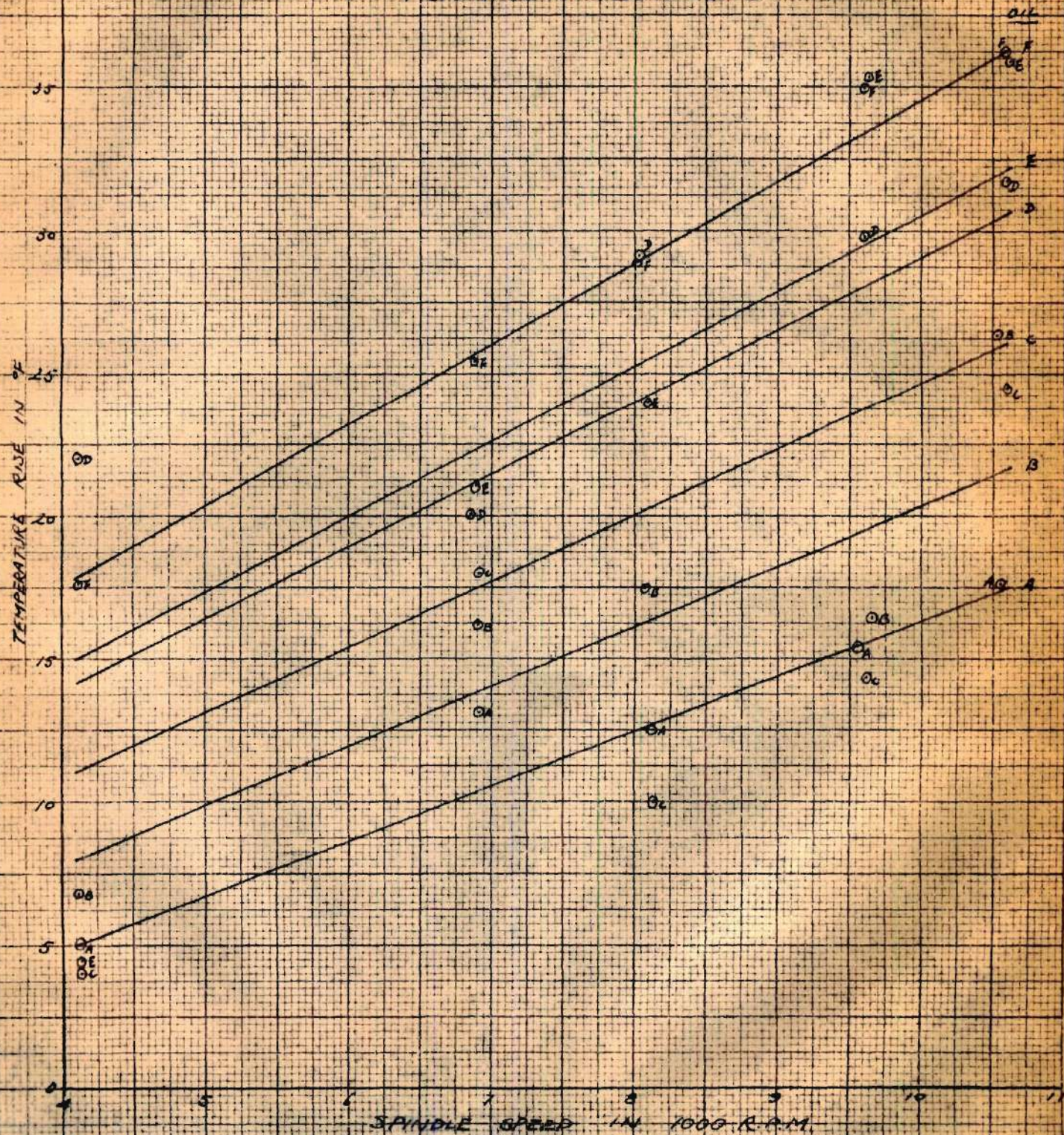
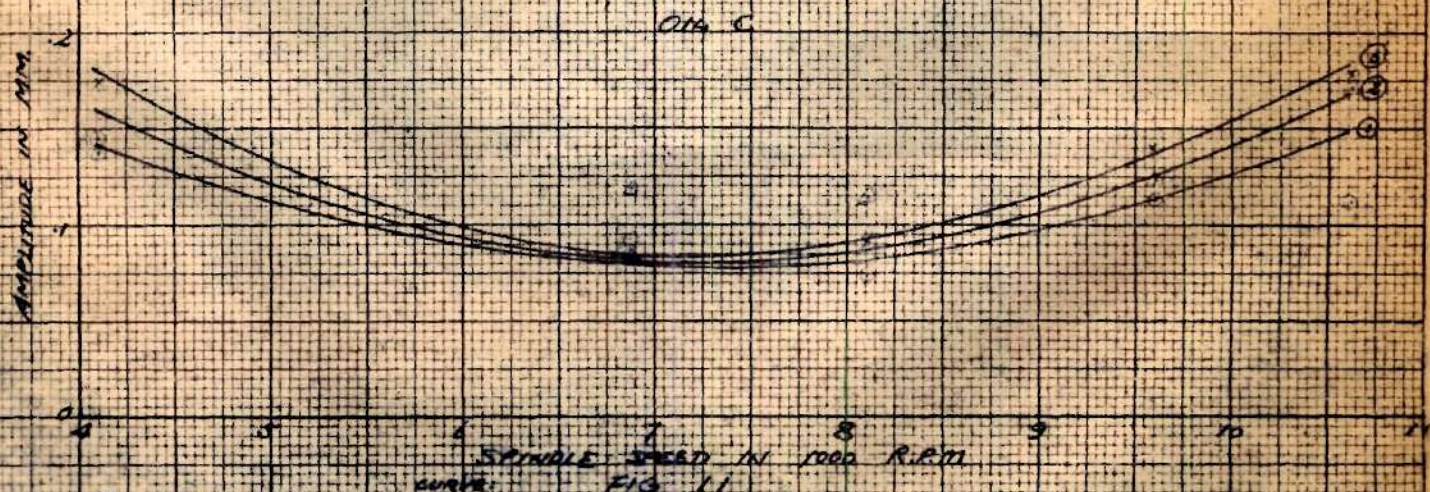
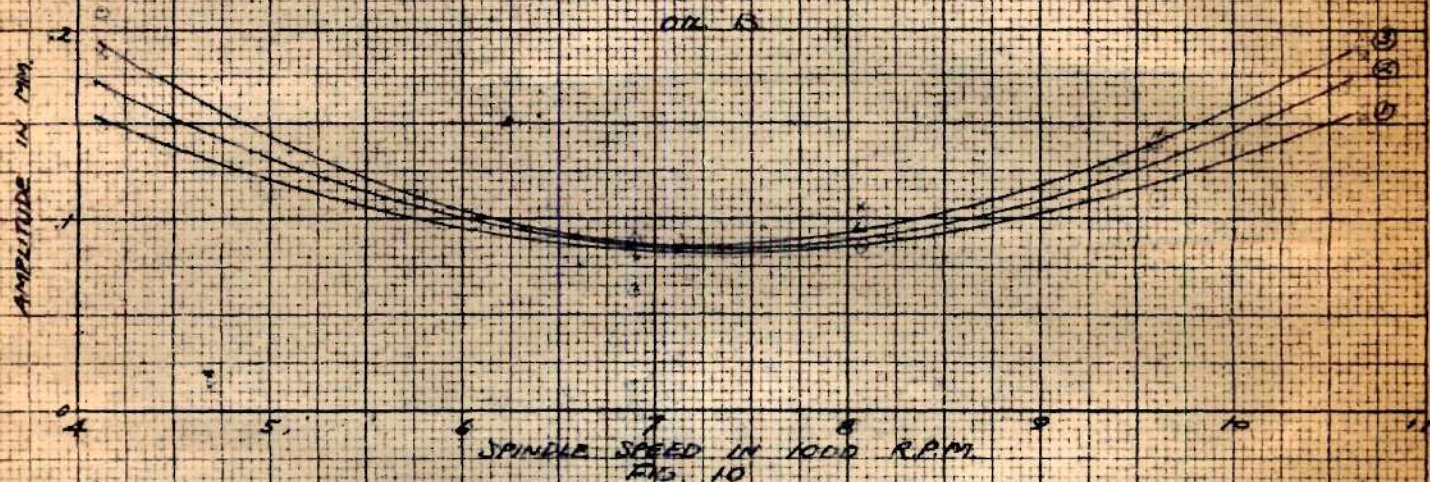
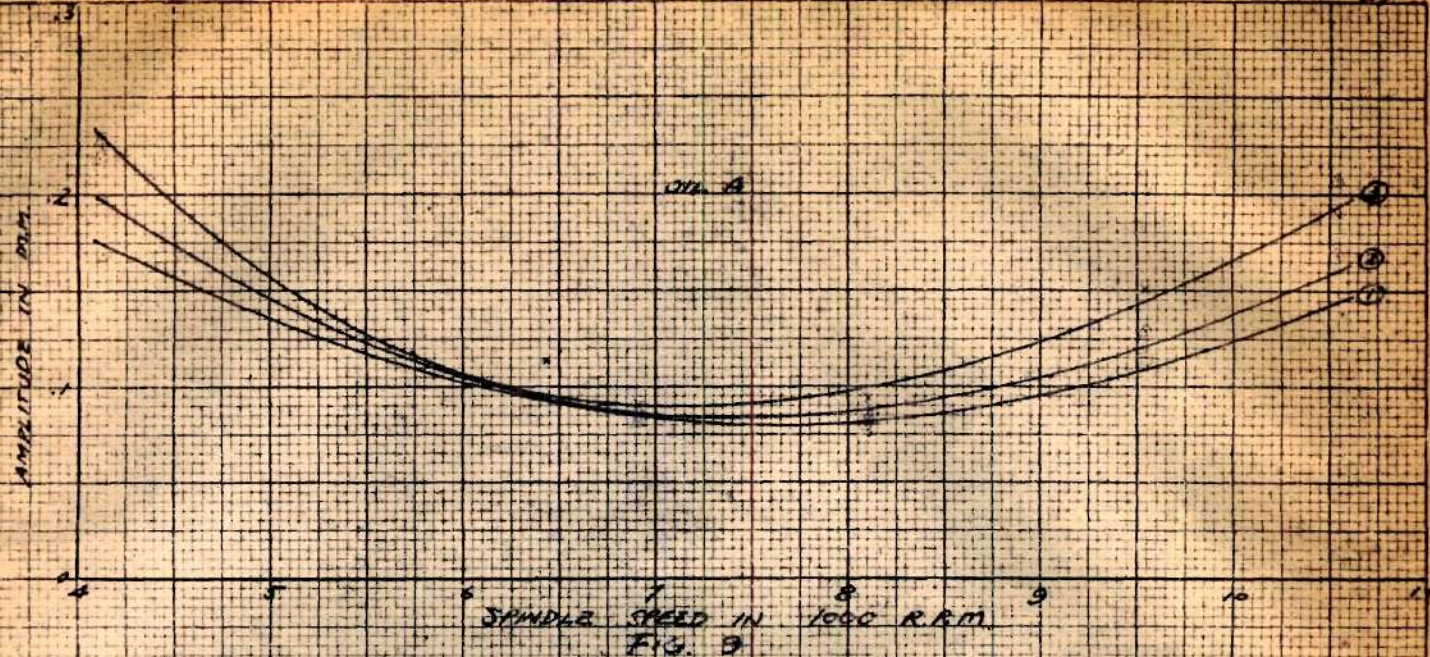


FIG. 5



- ① - EMPTY BOBWIN
 ② - 1/2 FULL BOBWIN
 ③ - FULL BOBWIN

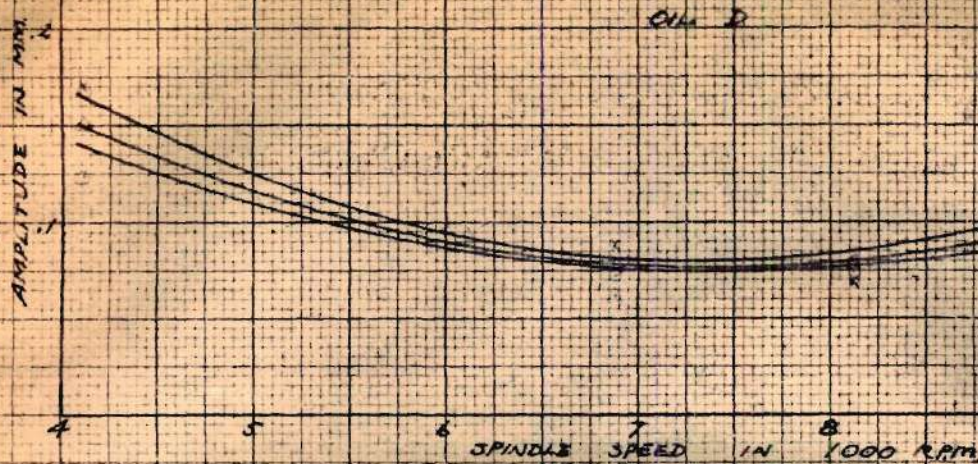


FIG. 12

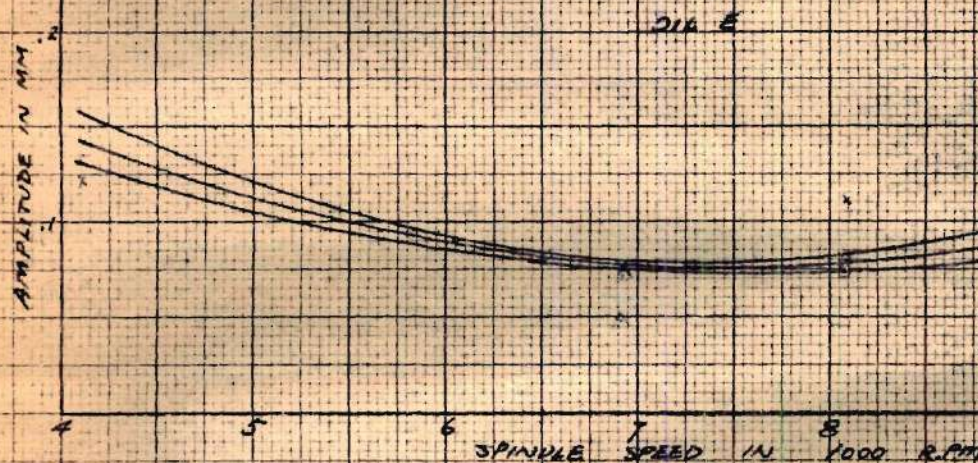


FIG. 13

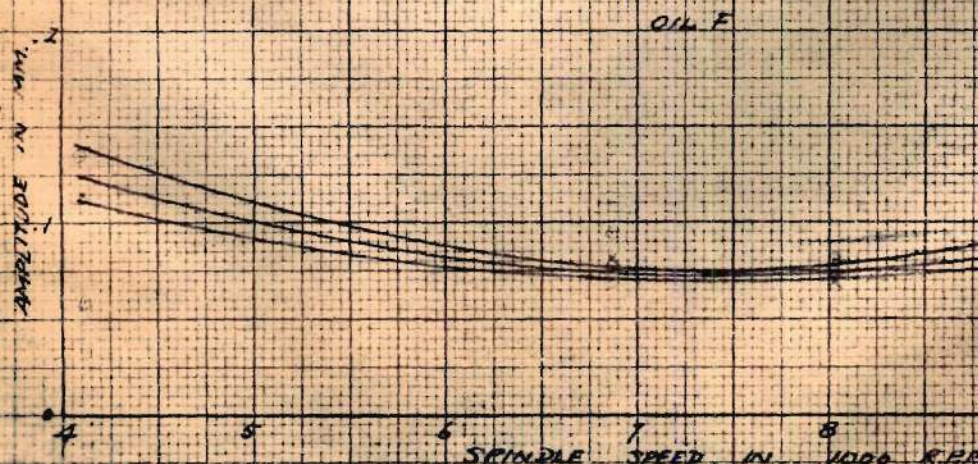


FIG. 14

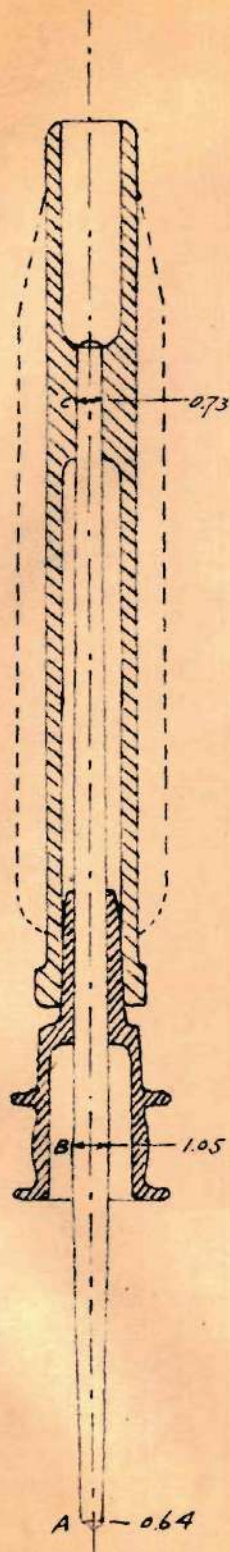
- CURVES
- - EMPTY BOBBIN - (1)
 - - 1/2 FULL BOBBIN - (2)
 - x - FULL BOBBIN - (3)

THEORETICAL CONSIDERATIONS

A cross-sectional view of the spindle and bobbin shown in Fig. 2 is shown in Fig. 15. This spindle has a weight of approximately 9 oz. and is supported at points A and B by the sleeve and footstep bearings. The bobbin weighs 2 oz. and is supported at C; at full bobbin, as shown in Fig. 15, the yarn weighs 2 oz. This small rotating body has brought about a problem of vibration due to the high speed of operation, not unlike that occurring in the operation of a steam turbine. The gyroscopic effect should be considered, also, in the analysis, since this is a case of "overhung disc" construction.⁴

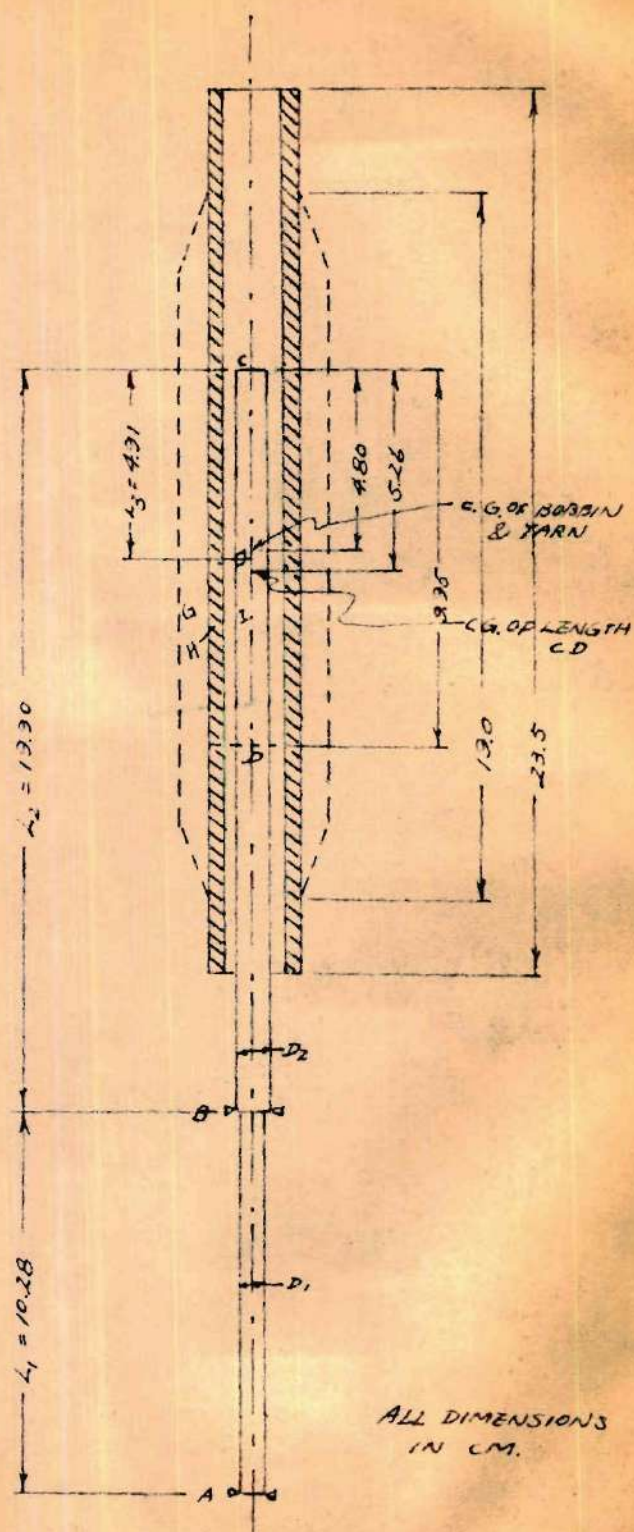
The assembly was redrawn in Fig. 16 to facilitate analysis. The spindle is assumed to be composed of two parts of different diameters D_1 and D_2 , and is supported at bearings A and B instead of the tapered bolster and footstep bearing shown in Fig. 2. It is also assumed that spindle portions AB and BC are flexible; that bobbin H does not offer resistance to deflection of spindle, and is supported at C with a comparatively long bearing surface; that AB has negligible mass; and that one half of BC from C,² bobbin H, and the yarn G have their center of gravity at O, having an eccentricity e (Fig. 17) from the geometrical axis. The whole mass may be

4. J. P. Den Hartog, "Gyroscopic Effect", Mechanical Vibrations, pp. 317-23, 1947. A. Stodola, "The Overhung Disc", Steam and Gas Turbine, Vol. 1, pp. 432, 1927.



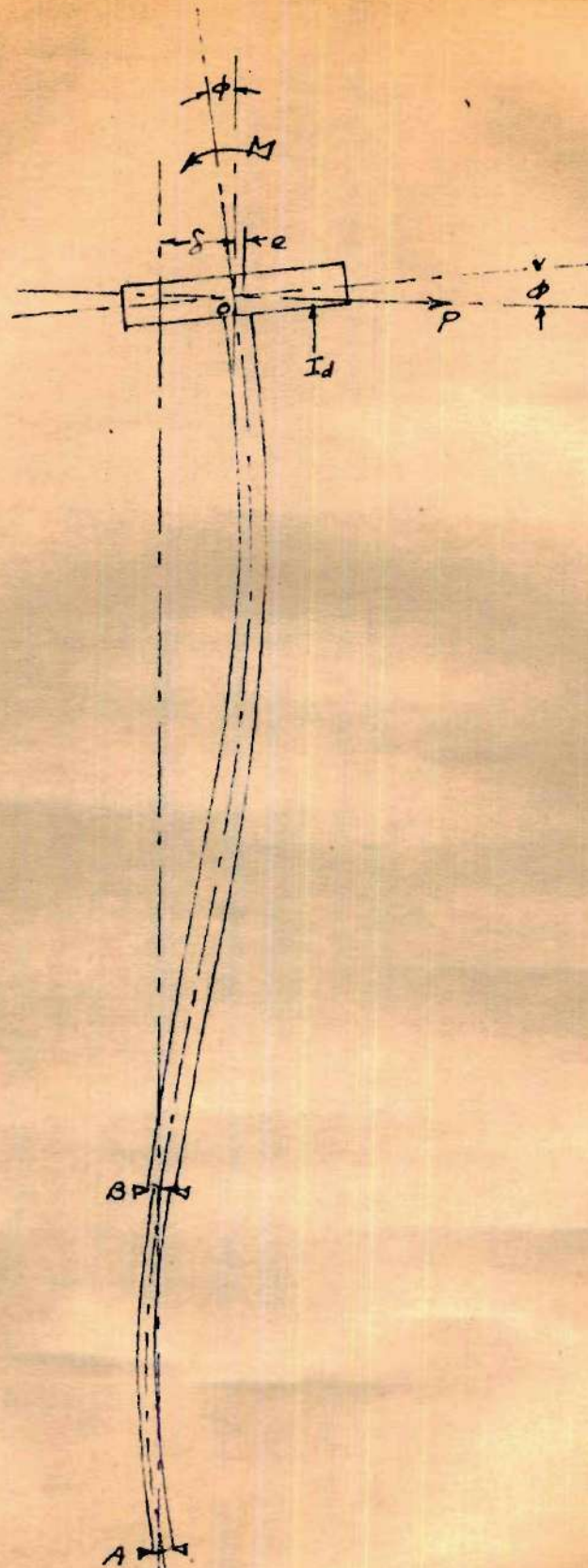
ACTUAL SPINDLE WITH
BOBBIN & YARN

FIG. 15



EQUIVALENT SPINDLE WITH
JOBBIN & YARN

FIG. 16



ACTION OF CENTRIFUGAL FORCE &
GYROSCOPIC MOMENT

FIG. 17

considered as a disc at O. The deflection curve in Fig. 17 shows the effect on deflection δ and angle ϕ made by the disc with respect to the horizontal plane due to the combined effect of centrifugal force P and gyroscopic moment M, which can be separated into δ_P and ϕ_P due to centrifugal force P, and δ_M and ϕ_M due to gyroscopic moment M.

$$P = m (\delta + e) \omega^2 = \text{centrifugal force at O,}$$

$$\text{and } M = I_d \phi \omega^2 = \text{gyroscopic moment about O,}$$

$$\text{where } m = m_G + m_H + m_I = \text{rotating mass at O,}$$

$$I_d = \text{moment of inertia of the body about one of its diameters,}$$

$$\delta = \text{deflection at O,}$$

$$\phi = \text{angle of the disc with respect to the horizontal plane,}$$

$$\text{and } \omega = \text{angular velocity.}$$

$$\text{Referring to Fig. 17, } \delta = \delta_P + \delta_M = p_1 P + p_2 M$$

$$\text{or } \delta = p_1 m (\delta + e) \omega^2 + p_2 I_d \phi \omega^2, \quad (1)$$

$$\text{and } \phi = \phi_P + \phi_M = q_1 P + q_2 M$$

$$\text{or } \phi = q_1 m (\delta + e) \omega^2 + q_2 I_d \phi \omega^2; \quad (2)$$

$$\begin{aligned} \text{where } p_1 &= \text{deflection per unit centrifugal force}^2 \\ &= \frac{1}{EI_2} \left(\frac{1}{3} L_2^3 + L_2 L_3^2 - L_2^2 L_3 \right) + \frac{1}{EI_1} \left(\frac{1}{3} L_1 L_2^2 + \frac{1}{3} L_1 L_3^2 \right. \\ &\quad \left. - \frac{2}{3} L_1 L_2 L_3 \right), \end{aligned}$$

$$p_2 = \text{deflection per unit gyroscopic moment}$$

$$= \text{angle of deflection per unit centrifugal force}$$

$$= -\frac{1}{EI_2}(\frac{1}{2}L_2^2 - L_2L_3) - \frac{1}{EI_1}(\frac{1}{3}L_1L_2 - \frac{1}{3}L_1L_3),$$

q_2 = angle of deflection per unit gyroscopic moment

$$= \frac{1}{EI_2}L_2 - \frac{1}{EI_1}\frac{L_1}{3};$$

I_1 = moment of inertia of cross-section $D_1 = \frac{\pi}{64}D_1^4$,

I_2 = moment of inertia of cross-section $D_2 = \frac{\pi}{64}D_2^4$,

E = modulus of elasticity of the material (steel),

and L_1, L_2, L_3 represent dimensions shown in Fig. 16.

Upon solving Equations (1) and (2) for δ ,

$$\delta = \frac{-e\omega^2[(p_1q_2 - p_2q_1)I_d m\omega^2 - p_1m]}{(p_1q_2 - p_2q_1)I_d m\omega^4 - (p_1m - q_2I_d)\omega^2 + 1}. \quad (3)$$

It can be seen that when the denominator of Equation (3) is equal to zero, i.e.,

$$(p_1q_2 - p_2q_1)I_d m\omega^4 - (p_1m - q_2I_d)\omega^2 + 1 = 0, \quad (4)$$

a resonance condition is established. Therefore, upon solving Equation (4), which is a quadratic equation, two values of ω_1 and ω_2 , namely, the first and second critical speeds, can be obtained because, from practical considerations, there are two real positive values for ω^2 which satisfy the equation. On the other hand, for minimum vibration, either ω^2 equals to zero, in which case the spindle is at rest; or

$$(p_1q_2 - p_2q_1)I_d m\omega^2 - p_1m = 0,$$

which, on being solved, gives:

$$\omega^2 = \frac{p_1}{(p_1q_2 - p_2q_1)I_d} = \omega_o'^2. \quad (5)$$

Equation (5) is the condition when the gyroscopic moment balances the moment due to centrifugal force. However, the absence of vibration of point O at ω_0' , theoretically, is a state of unstable equilibrium; thus, having very little significance. Equation (3) can be written as:

$$\delta = \frac{-e\omega^2(\omega^2 - \omega_0'^2)}{(\omega^2 - \omega_1^2)(\omega^2 - \omega_2^2)} \quad (6)$$

From Equation (6) the resonance diagram can be plotted as shown in Fig. 18(a) in which the phase relation is clearly shown. The shapes of spindle vibration are shown in Fig. 18(b) as interpreted from the resonance diagram. Yet, the diagram shown in Fig. 18(a) is the resonance diagram for the "absolute motion" which cannot be measured by the kind of apparatus used in this experiment. Since the amplitude of vibration is of main interest in this study, the phase relation may be disregarded and the resonance diagram can be replotted as in Fig. 18(c) in which the "relative motion" of the system is shown; the values of amplitude represented by the ordinates being those which can possibly be measured.⁵ This also explains the continuity of the resonance curve from actual measurement since the deflection, even at critical speed, remains finite due to friction.⁶ The minimum amplitude of vibration between

5. J. P. Den Hartog, "Undamped Forced Vibrations", Mechanical Vibrations, pp. 59-61, 75, 1947.

6. A. Stodola, "Effect of Resistance to motion", Steam and Gas Turbine, Vol. 1, pp. 428-9, 1927.

the first and second critical speeds occurs when $\delta = -e$, which is the condition that occurs when the center of gravity coincides with the vertical axis; i.e.,

$$-e = \frac{-e \omega^2 [(p_1 q_2 - p_2 q_1) I_d m \omega^2 - p_1 m]}{1 - (p_1 m - q_2 I_d) \omega^2 + (p_1 q_2 - p_2 q_1) I_d m \omega^4},$$

which, upon being solved, gives $\omega^2 = \frac{1}{q_2 I_d} = \omega_0^2$. (7)

The following calculation, using the values listed below, is made on the system shown in Figs. 15 and 16 to obtain the numerical values which are compared with experimental data in the following section. All dimensions are expressed in c.g.s. system for simplicity in calculation.

$$D_2 = \frac{1}{2} (0.73 + 1.05) = 0.89 \text{ cm.}$$

$$D_1 = \frac{1}{2} (0.90 + 0.64) = 0.77 \text{ cm.}$$

$$I_2 = \frac{\pi}{64} 0.89^4 = 30.4 \times 10^{-3} \text{ cm.}^4$$

$$I_1 = \frac{\pi}{64} 0.77^4 = 17.0 \times 10^{-3} \text{ cm.}^4$$

$$E = 2.06 \times 10^{12} \text{ gms./cm.}^2 \text{ for steel.}$$

$$m_G = 65.90 \text{ gms., } m_H = 54.35 \text{ gms., } m_I = 36.5 \text{ gms.}$$

$$m = m_G + m_H + m_I = 156.8 \text{ gms.}$$

$$I_d = 4765 \text{ gms.-cm.}^2$$

$$L_1 = 10.28 \text{ cm., } L_2 = 19.90 \text{ cm., } L_3 = 4.91 \text{ cm.}$$

$$p_1 = 40.41 \times 10^{-9}, p_2 = q_1 = -3.052 \times 10^{-9},$$

$$q_2 = 0.416 \times 10^{-9}.$$

Upon substituting the above values into Equation (4),

$$5.6 \times 10^{-12} \omega^4 - 8.668 \times 10^{-6} \omega^2 + 1 = 0,$$

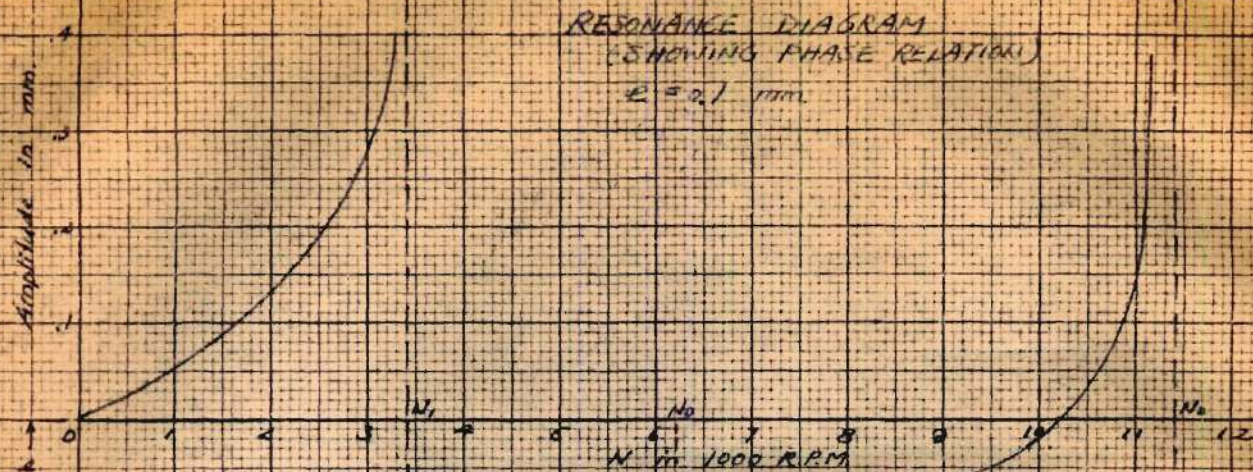
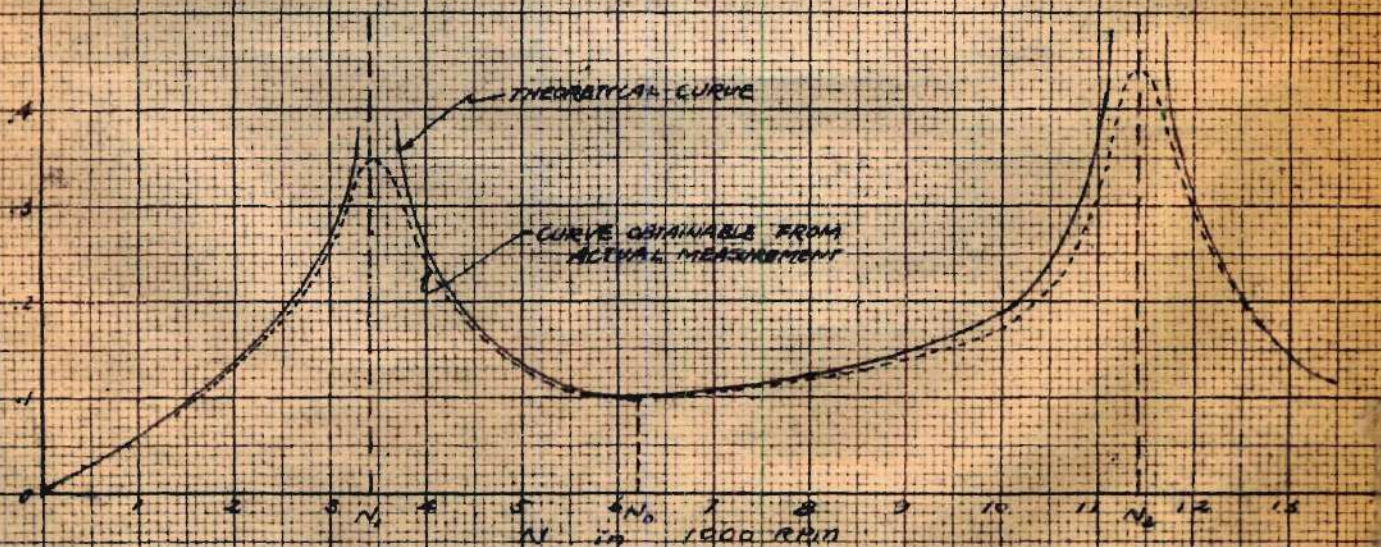


Fig 18 (a)

Fig 18 (b) SHAPES OF VIBRATION

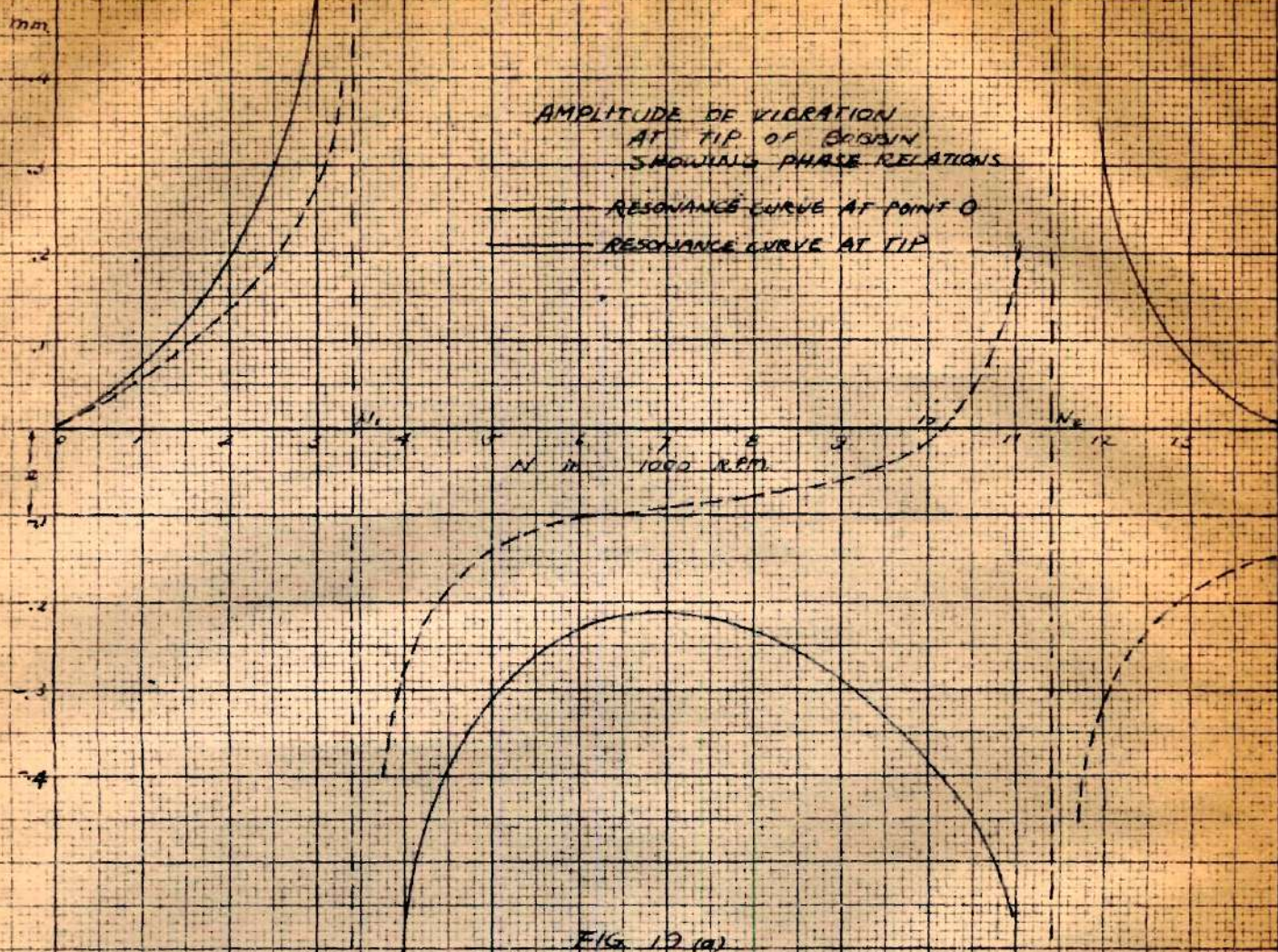


RESONANCE DIAGRAM $\zeta = 2.1 \text{ mm}$
(SHOWING AMPLITUDE ONLY)
Fig 18 (c)

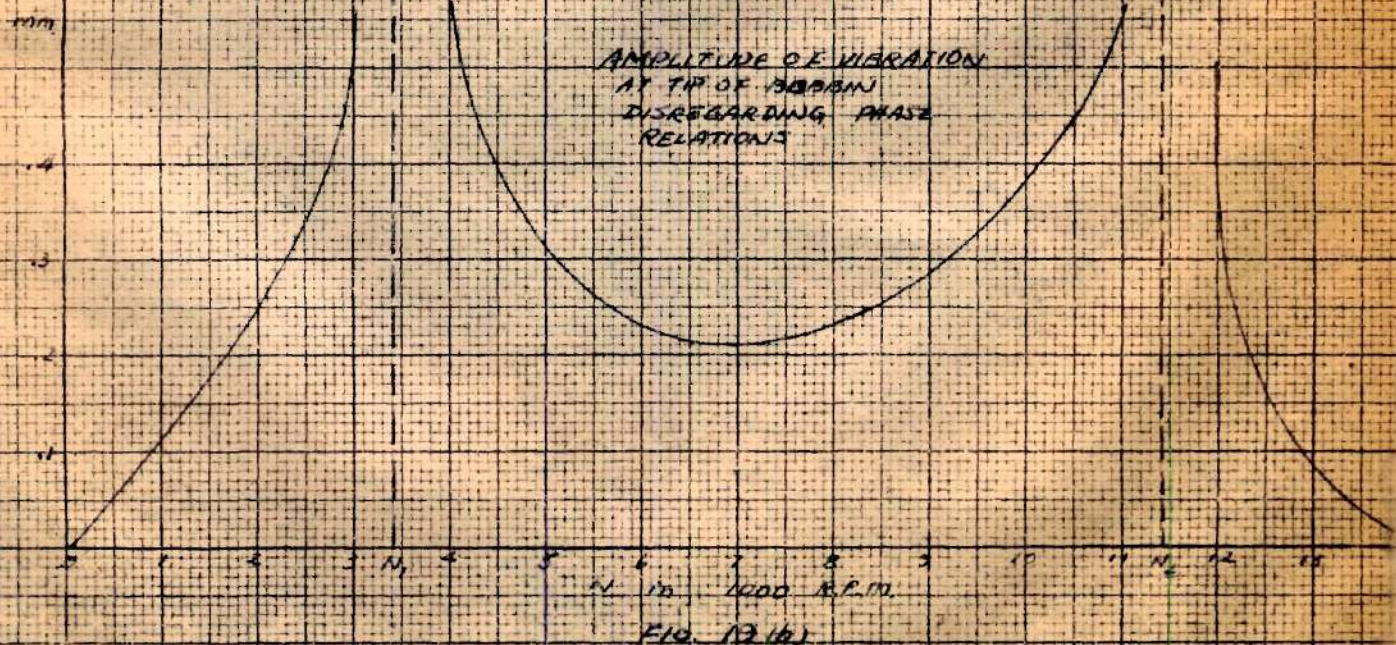
AMPLITUDE OF VIBRATION
AT TIP OF BOBBIN
SHOWING PHASE RELATIONS

RESONANCE CURVE AT POINT O

RESONANCE CURVE AT TIP



AMPLITUDE OF VIBRATION
AT TIP OF BOBBIN
DISREGARDING PHASE
RELATIONS



which, upon being solved yields

$$\omega^2 = 0.1265 \times 10^6 \quad \text{or} \quad 1.421 \times 10^6,$$

$$\omega_1 = 356 \text{ rad./sec.}, \quad \omega_2 = 1194 \text{ rad./sec.},$$

$$N_1 = 3042 \text{ R.P.M.}, \quad \text{and} \quad N_2 = 11,420 \text{ R.P.M.}$$

Also, from Equation (5), $\omega_0'^2 = 1.132 \times 10^6$,

$$\omega_0' = 1064 \text{ rad./sec.}, \quad \text{and} \quad N_0' = 10,170 \text{ R.P.M.};$$

and, from Equation (7), $\omega_0^2 = 0.422 \times 10^6$,

$$\omega_0 = 650 \text{ rad./sec.}, \quad \text{and} \quad N_0 = 6210 \text{ R.P.M.}$$

The equation representing the resonance curve is:

$$\delta = \frac{e \omega^2 (\omega^2 - 1.132 \times 10^6)}{(\omega^2 - 0.1265 \times 10^6)(\omega^2 - 1.421 \times 10^6)} \quad (8)$$

The diagram plotted is shown in Fig. 18(a), assuming $e = 0.10$ mm., and is rectified as shown in Fig. 18(c).

As the measurement of amplitude of vibration was made at the tip of the bobbin, which is 11.0 cm. from 0, the center of gravity of the system, further analysis is necessary to determine the resonance curve at the tip; this data being desired for convenience of comparison in the next section. It can be deduced from Fig. 17 that:

$$D = \text{deflection at tip} = \delta - \phi L, \quad (9)$$

where L = distance from 0 to tip = 11.0 cm.

From Equation (2),

$$\phi = \frac{q_1 m (\delta + e) \omega^2}{1 - q_2 I_d \omega^2}$$

$$= \frac{-0.478 \times 10^{-6} (\delta + e) \omega^2}{1 - 2.328 \times 10^{-6} \omega^2} . \quad (10)$$

The diagram shown in Fig. 18(a) is plotted again in Fig. 19(a) in dotted line with the value of ϕL superimposed vectorially as shown in full line. Neglecting the phase relations, the resonance diagram shown in Fig. 19(b) represents the theoretical value of amplitude that can be measured at the tip.

DISCUSSION

Factors affecting vibration. To facilitate discussion, all the following factors affecting vibration should be considered: ⁷

1. The oil in the spindle base outside the bolster acts as a shock absorber to keep the vibration of the spindle from being transmitted only to the bolster rail, and the slight movement of the bolster permitted by this construction does not in any way affect vibration of the spindle itself under steady condition of operation.

2. The bobbin is attached to the spindle blade by a push fit, yet due to the small mass of bobbin and yarn (4 oz. in this experiment) as compared with the whole spindle (9 oz.), the internal hysteresis of this system is not sufficient to excite whirling even if slippage occurs, and any slippage tends to damp down torsional vibration only.

3. Since there is a copious supply of oil due to bath lubrication, oil whip does affect the amplitude of linear vibration.

4. The pull of the driving tape is at the center of upper bearing and, therefore, does not affect linear vibration.

7. J. P. Den Hartog, "Internal Hysteresis of Shafts and Oil-film Lubrication in Bearings as causes of Instability", Mechanical Vibrations, pp. 361-6, 1947. A. L. Kimball, "Shaft Whirling due to Internal Friction and Oil Action in Journal Bearing", Vibration Prevention in Engineering, pp.124-9, 1932.

Slippage of the driving tape around the whorl tends to damp down the torsional vibration as a case of "Friction Damper".

5. Due to the effective damping towards torsional vibration, it is not considered in this study; besides, even large torsional vibration does not affect average power consumption of the spindle.

Therefore, the only factor which tends to affect the amplitude of linear vibration is the oil whip. Although oil whip is not a damping or friction phenomenon, dry friction due to incomplete lubrication has been proved to be sure to court bad whirling. The net effect due to complete lubrication remains to be determined for each individual case. The so-called "wobble"⁸ is in reality the mechanical vibration either excited or damped by the action of oil. The power consumption is affected by the linear vibration because the bearing pressure is affected by the deflection.

Vibration as affected by oil. While the apparatus used in this work lacks refinement, it does provide a means for explaining the fundamental theories. To show the effect of lubrication on vibration, the amplitudes of vibration of the tip at full bobbin that occurred when various lubricants were used, were converted to the amplitude of point O as

8. F. S. Jones and S. P. Marley, "Textile Spindle Performance Characteristics", presented at annual meeting of A. S. L. E., 1948.

shown in Fig. 16. The ratio of the amplitude of O at a given speed, as shown in Fig. 18(c), to the amplitude of the tip at the same speed, as shown in Fig. 19(b), was used as the conversion factor at each speed. The estimated values were plotted in Fig. 20 together with the calculated resonance curve for an eccentricity of 0.053 mm. which is shown by the dotted line. Although the curves are not sufficiently refined to obtain such numerical values as damping constants,⁹ it can be seen that the damping effect is also variable for same oil at different speeds and different operating temperatures (Fig. 8). By comparison of the amplitudes shown in Fig. 20, it can be seen that the damping effect of the heavier oils toward vibration is more pronounced when the spindle is operated near the critical speeds, while at the speed near the speed of minimum amplitude of vibration, the damping effect is very slight. Similar comparisons can be made on the amplitude of vibration when the spindle is operated at half full and empty bobbin. As vibration can be expected to decrease as the rotating mass becomes smaller, these conditions were not included in this investigation.

Power consumption and oil viscosity. The series of curves in Fig. 6 on page 12 shows that power consumption

9. J. P. Den Hartog, "Free Vibration with Viscous Damping", Mechanical Vibrations, pp. 51-7, 1947.

increases when a heavier oil is used, as compared with that when a lighter oil is used. This phenomenon is substantiated by the higher bearing oil temperature when a heavier oil is used instead of a lighter oil (Fig. 8) which means larger quantity of heat generated by friction and therefore a waste in power, and the higher percentage slip of tape drive when a heavier oil is used due to greater fluid resistance as shown in Fig. 7. The increased fluid friction of a heavier oil, of course, means a greater friction torque to rotation, but, also, reduces the efficiency of the tape drive, thus making the power consumption still higher. For the same oil, percentage slip is higher when the spindle is operated at a low speed than at a high speed, thus the driving efficiency was less at low speeds. Without considering these factors, for film lubrication, which was the operating condition throughout this study, power consumption is proportional to the square of the operating speed.¹⁰ However, this statement is based on the condition that the lubricating oil is assumed to be a "Newtonian liquid" which is far from the actual case, as all oil thins out at the temperatures resulting from the use of relatively high speeds. Therefore, a power-speed curve of such a shape that has less curvature than a parabola may be assumed if the increase in bearing pressure due to larger de-

10. J. L. Clower, "Fundamentals of Lubrication", Lubricants and Lubrication, pp. 163-8, 1939.

flection be neglected. In the study of some authors, the power-speed curve is a straight line.⁸ The actual case should be something intermediate between the two. Yet, it can be seen in Fig. 6 that the curvature of the power-speed curves of different oils is even greater than that of a parabola. The only factor that can be responsible for this fact is the different amplitude of vibration at different speed which affects the bearing pressure which is increased by increased deflection. The power-speed curve of oil A in Fig. 6 is redrawn in Fig. 21, together with a parabola and a straight line tangent to this curve. The parabola is represented mathematically by the equation:

$$P = 0.1055 \times 10^{-6} N^2,$$

where P is the power in watts and N is the R.P.M. of the spindle. As stated previously, without vibration the experimental curve should be intermediate between the parabola and the straight line. To be conservative, assume the straight line relation be the case. The difference in ordinate between the experimental curve and the straight line, shown shaded in Fig. 21, is the power consumed due to linear vibration. Of course, this method of analysis has very questionable accuracy; however, it is very interesting to notice the similarity of the curves of power consumed due to vibration as shown in Fig. 22 to the resonance curves shown in Fig. 20. Due to larger fluid friction in the case of heavier oil than that of

ESTIMATED AMPLITUDE OF VIBRATION
AT POINT O IN FIG. 16
(FULL GROUND)

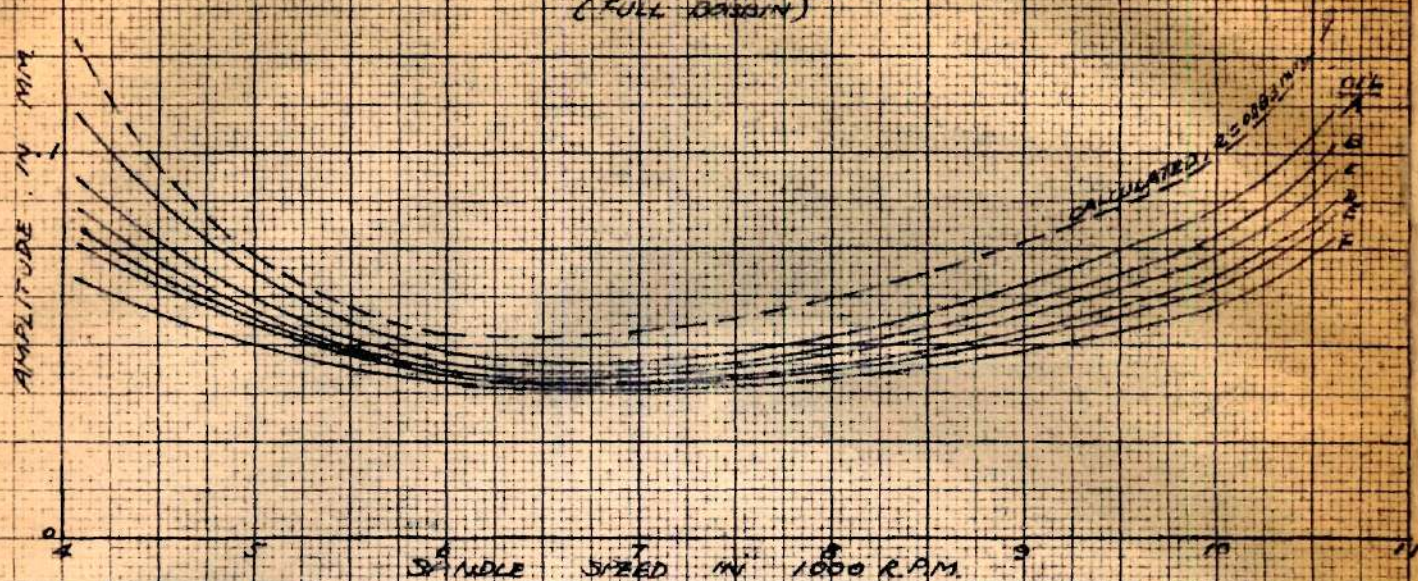


FIG. 20

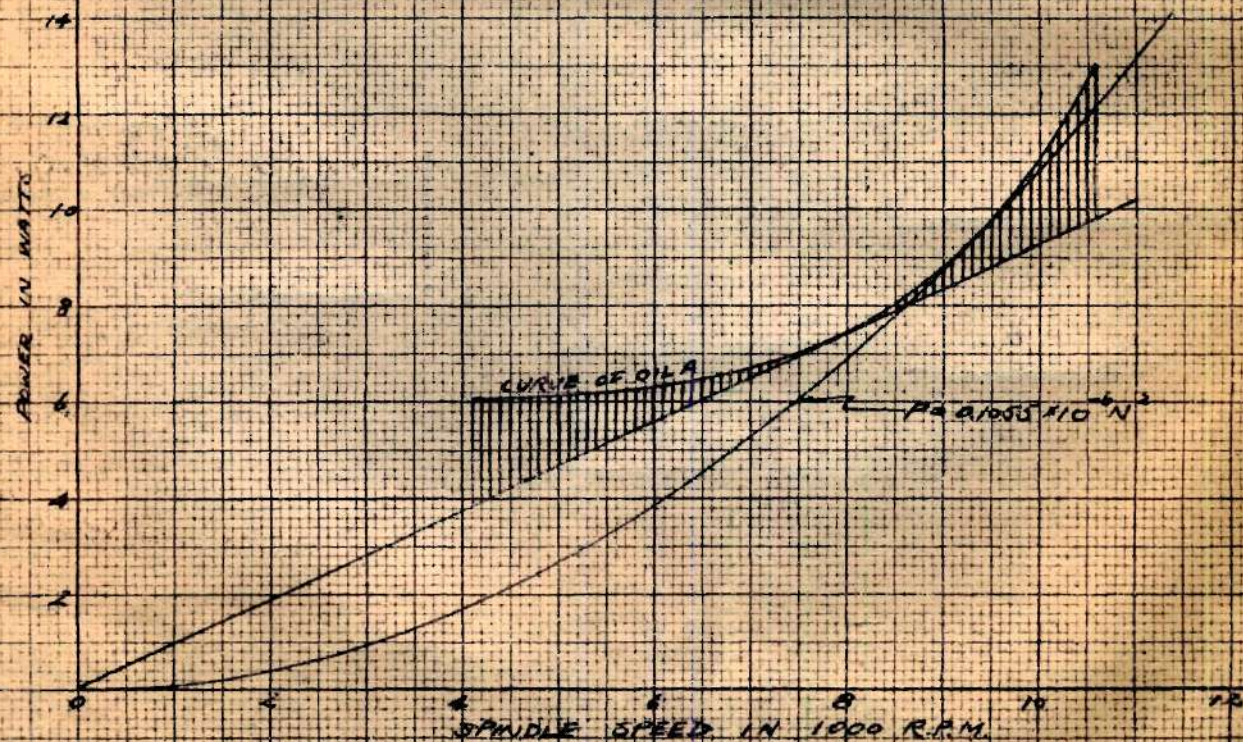
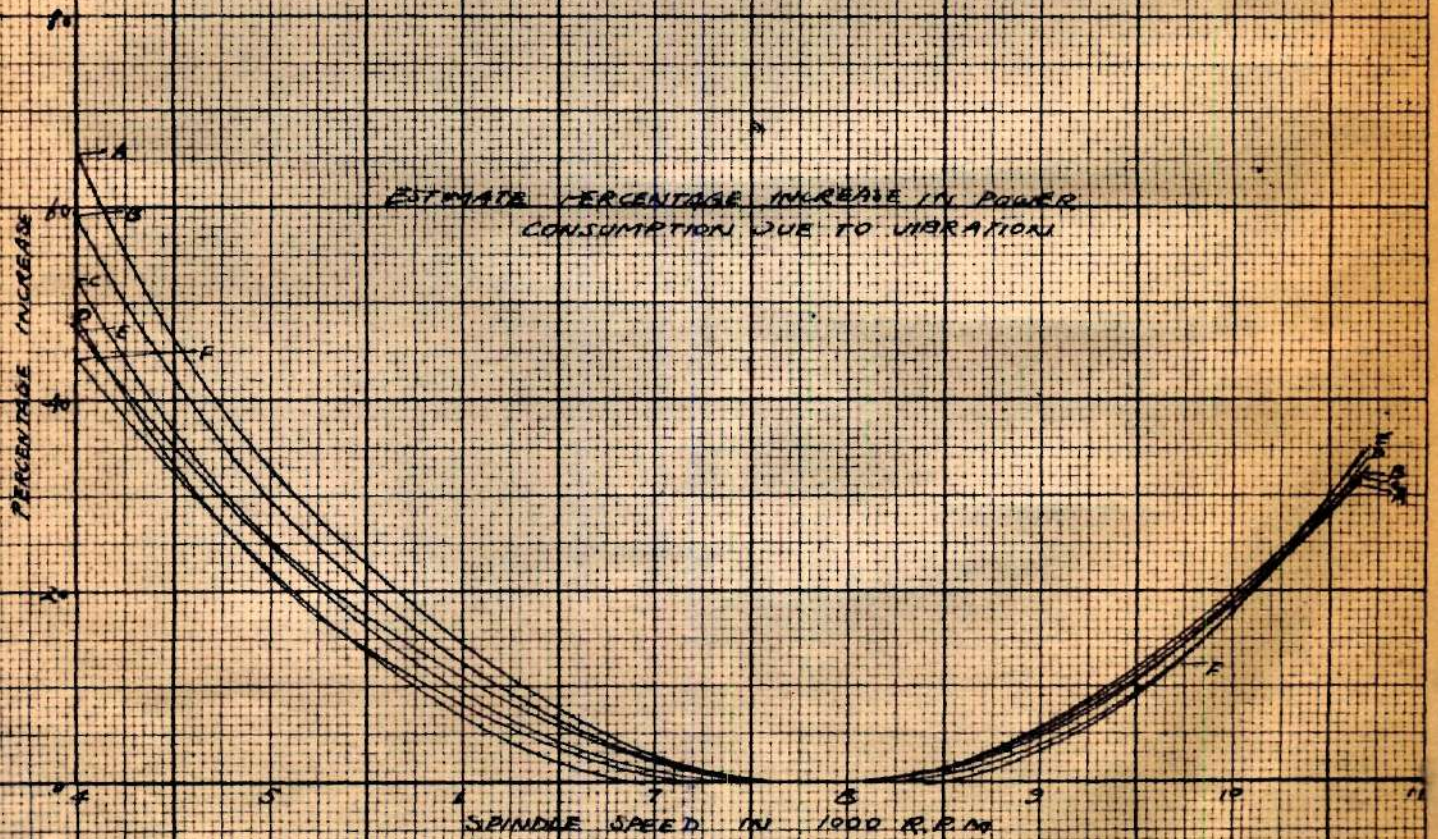
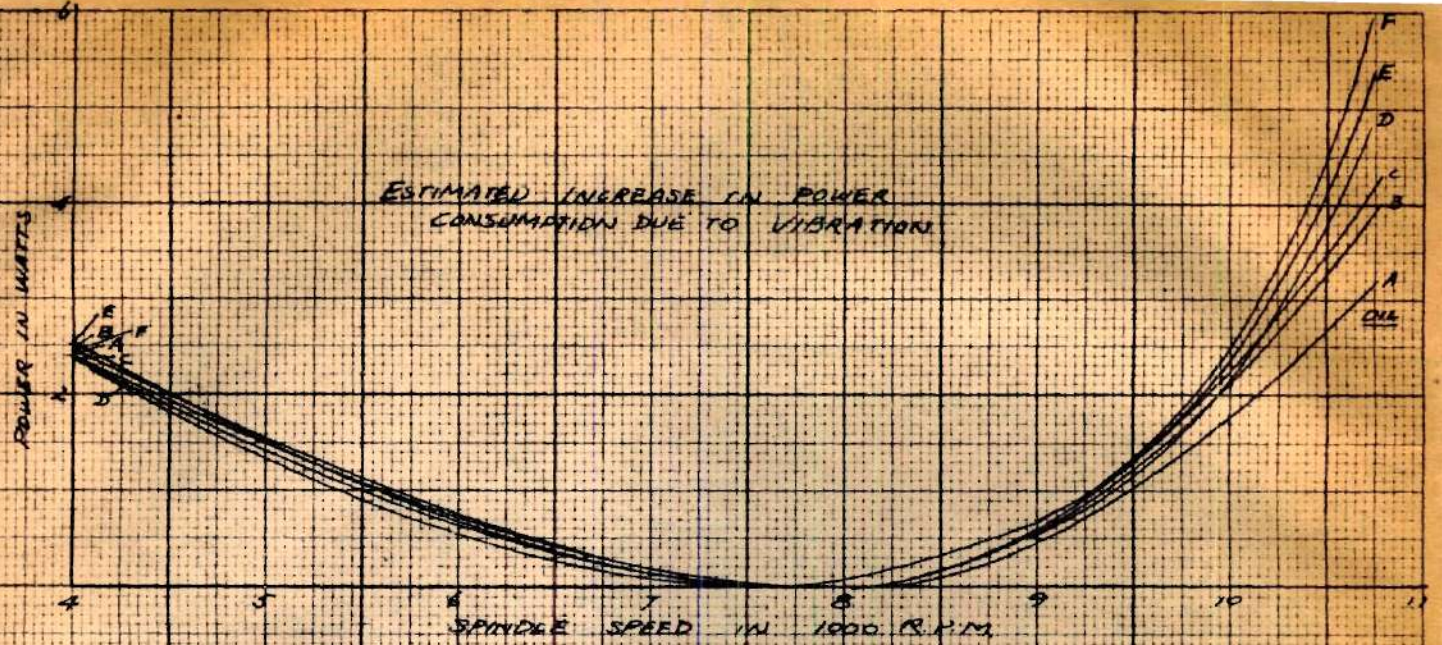


FIG. 21



lighter oil, the power consumed due to vibration in the case of heavier oil which has less amplitude as shown in Fig. 20 is more than that of the lighter oil. The curves of percentage increase in power consumption from the straight line relation which are shown in Fig. 23 explains the fact that while the amount of increase is larger in the case of heavier oil, the percentage increase, on the whole, is less due to damping characteristics of heavier oil. It is to be remembered that the curves in Figs. 22 and 23 only serve to assist in interpreting the theory regarding power consumption and their numerical values can by no means be considered as having any significance.

Another fact that is worthy of consideration is the large percentage of power required to drive the spindle as compared with other machine parts of the spinning frame (Fig. 5). The spinning frame used in this experiment has only 36 spindles. With spindles driven at 4,000 R.P.M., using spindle oil A, 55% of the 0.53 total horsepower was required to drive the spindles; at 9,000 R.P.M., 47.5% of 0.86 total horsepower was required; while in the case of a regular 256-spindle frame, at 4,000 R.P.M., 90% of the $2\frac{1}{2}$ total horsepower, estimated according to the curves of power distribution as shown in Fig. 5, will be consumed in spindle drive, and at 9,000 R.P.M., 87% of the $3\frac{1}{2}$ horsepower estimated will be consumed. However, using spindle oil C, for a 256-spindle frame,

at 4,000 R. P.M., 91% of 2.6 total horsepower estimated will be required in spindle drive, and at 9,000 R.P.M., 89% of 4.2 total horsepower estimated will be required in spindle drive. This illustration shows the importance of selecting proper oil for spindle lubrication which affects greatly the power consumption of the spinning frame and, thereby, the power consumption of the spinning department, which, in turn, requires the largest percentage of power of a whole mill.¹¹

Oil throwing, though not serious in staining the yarn, wastes oil and tends to make the floor slippery. It should be obviated, not only for the sake of obtaining perfection, but for the reason that straight mineral oil, which has been found to be the only practical substance for use in lubricating spindles, cannot be removed from yarn in finishing process.¹² Oil throwing is mainly a result of using more oil than is necessary for lubrication and the height of the spindle base can retain under operating conditions.

11. "Textile Machinery Lubrication", Lubrication, No. 2, Vol. 16, pp. 13-24, 1930.

12. "Spindle Design and Function of Spindle Oils", Lubrication, No. 3, Vol. 23, pp. 25-31, 1937.

CONCLUSIONS

From the results of this study, it is concluded that:

1. There is a narrow optimum speed range for minimum mechanical vibration for each design of spindle; the presumption of Y. Sakurai that a spindle should be operated at speeds $\pm 15\%$ away from its critical speed ² is necessary but not sufficient.

2. Linear vibration of a spindle is affected by the lubricating oil used; greater damping effect can be expected by employing heavier oil when operating at near critical speeds, yet the damping effect is not obvious when operating near optimum speed.

3. Power consumption for driving a spindle is much greater when heavier oil is used, hence, all aspects of introducing a heavier oil for the purpose of damping down vibration should be considered.

4. Power consumption for driving a spindle is greater when operating near critical speed than it would be if vibration were absent.

5. Oil throwing is mainly the result of using more oil than is necessary.

RECOMMENDATIONS

Since it is impossible to manufacture a spindle or bobbin of zero eccentricity, it is recommended that in addition to checking for the critical speeds, adjustment of the shape should be made so that the designed speed be the optimum operating speed. The use of heavier oil for operating near critical speed is an expensive project and should be avoided. Therefore economical operation can be obtained by operating the spindle at optimum operating speed using the lightest oil just sufficient for maintaining film lubrication.

The height of the oil tube should be just high enough to retain sufficient oil for lubrication, yet not to cause oil throwing. The location of the doffer guard should be a secondary consideration.

Although this study was made on sleeve bearing spindles for cotton spinning, similar studies can be made on spinning spindles for other fibers and for twister spindles, as the principle of operation remains the same. The principle of lubrication is different in case of ball or roller bearing spindles; however, similar analysis as to design for the optimum operating speed would be advantageous regarding smooth operation and power saving.

This study has produced an idea for the development of a single standard oil for spindle lubrication. With larger or smaller packages which require larger or smaller driving

power and thereby impose larger or smaller forces at the bearing, the diameter of the journal of the spindle can be altered, not for the sake of strength and rigidity of the spindle alone, but with due consideration of bearing pressure to maintain sufficient oil film when that particular standard spindle oil is used. This endeavor should of course be made by spindle manufacturer. The result obtained may lead to more efficient and simplified spinning room operation.

It is also recommended that further work be done with spindles of various sizes. In further studies of the vibration, the use of two micrometers mounted on a calibrated yoke with two different electrical systems will give more accurate results. Also, a seamless collar for the purpose of grounding electric current should be mounted on the bobbin for obtaining better accuracy.



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APPENDIX

TABLE II

POWER DISTRIBUTION ON
THE SPINNING FRAME, IN H. P.

Oil Procedure	Average spindle speed, in R.P.M.				
	4,100	6,900	8,100	9,600	10,600
A	(1) 0.176	0.282	0.330	0.389	0.428
	(2) 0.478	0.571	0.826	0.879	1.042
	(3) 0.513	0.628	0.896	0.947	1.124
	(4) 0.035	0.057	0.070	0.068	0.082
	(5) 0.302	0.289	0.496	0.490	0.614
B	(2) 0.473	0.611	0.699	1.021	1.222
	(3) 0.517	0.666	0.766	1.094	1.239
	(4) 0.044	0.055	0.067	0.073	0.017
	(5) 0.297	0.323	0.369	0.632	0.794
	(2) 0.515	0.667	0.946	1.086	1.142
C	(3) 0.545	0.729	1.012	1.161	1.233
	(4) 0.030	0.062	0.066	0.075	0.091
	(5) 0.339	0.385	0.616	0.697	0.714
	(2) 0.544	0.697	1.010	1.087	1.187
	(3) 0.582	0.755	1.079	1.161	1.263
D	(4) 0.038	0.058	0.069	0.074	0.076
	(5) 0.368	0.405	0.680	0.698	0.759
	(2) 0.553	0.750	0.866	1.139	1.175
	(3) 0.585	0.804	0.939	1.210	1.260
	(4) 0.032	0.054	0.073	0.071	0.085
E	(5) 0.377	0.468	0.536	0.750	0.747
	(2) 0.559	0.731	1.022	1.150	1.357
	(3) 0.590	0.791	1.085	1.222	1.445
	(4) 0.031	0.060	0.063	0.072	0.088
	(5) 0.383	0.449	0.682	0.761	0.929
F	(2) 0.559	0.731	1.022	1.150	1.357
	(3) 0.590	0.791	1.085	1.222	1.445
	(4) 0.031	0.060	0.063	0.072	0.088
	(5) 0.383	0.449	0.682	0.761	0.929

Note: (1) Horsepower for drafting, with tapes detached.
 (2) Horsepower for drafting and driving spindles but without spinning.
 (3) Total horsepower, with spinning and drafting.
 (4) Horsepower for spinning only ((3) - (2)).
 (5) Horsepower required for driving spindles only ((2) - (1)).

TABLE III
EQUIVALENT OILS BY DIFFERENT PRODUCERS
FOR SPINDLE LUBRICATION ¹³

Viscosity, S.U.V. at 100°F		50-60	70-80	100-110	140-160	220-225
A. S. M. E. No.		5	6	7	8	9
Producer	Brand					
Socony Vacuum	Gargoyle Velocite	E	S	D	B	Vacuoline oil Medium
Sinclair	Lily White	D	A	B	Sintur-lite oil	Cordymo
N.Y. and N.J. Lubricant Co.	Non-fluid oil N.F.O. Spindle oil	1194	124	154	1150	2002
Eiske	Spindle oil	L	LM	M	H	XH
Harris	Spindle oil	Extra light	Gravity	Light	H.S.	Heavy
Houghton	Kensington Spindle oil	5	7	9	11	MIH-10
Texaco	Spindura	AA	BB	CC	EE	FF
Cities Service	Grade No.	1	2	4	Pacemaker No. 1	Pacemaker No. 2
Standard Oil Co. of Indiana	Indoil	5	7	9	15	25

¹³. "Correct Lubrication Essential to Production Efficiency", Rayon Textile Monthly, No. 3, Vol. 26, pp. 131-3, 1945.